CHAPTER 4

Other Loads

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CHAPTER 4

Other Loads

Introduction

1 The loads covered in this Chapter are listed on the contents page (4.1). Specifically, they consist of the most important loads which may be met by a ship in normal operation and the most common accident or wartime conditions but which are not due to wave effects. The order in which they will be discussed starts with the remaining environmental loads, such as wind and ice, then operational and accidental loads and finally loads due to weapon effects. Estimates to some degree of accuracy will be provided for all the load effects but some of them are essentially arbitrary and in these areas there is need for more research or at least empirical study.

2 Whether the loads act independently of each other, or have to be taken in conjunction with other loads, has to be left in the main to common sense engineering judgement but guidance will be given where possible.

Wind Loads

3 The wind acting on the above water part of the hull creates forces which the designer may need to consider. However, the majority of the above water structure will be designed to withstand green sea loads (Chapter 3); clearly wind and sea loads cannot act simultaneously at the same point and green sea loads will always be greater than wind loads for the same design condition. Consequently, for the majority of the structure the effect of wind may be ignored. Wind is usually only of significance in relation to steady forces which affect stability and propulsion resistance and which are not the subject of this book, but there are a few parts of ships in which wind forces affect structural design, for example on masts, and some advice will therefore be provided here.

4 Because pressures are relatively low there is little published data on the *local* effects of wind on marine structures, and most information available relates to the effect of overall wind loads on ship motion, for example as described by Gould (1982). However, in relation to overall resistance, Rawson and Tupper (1976) suggest a pressure not exceeding $0.2\ V^2$ Pa where the relative wind speed V is in m/s. Reference to Defence Standard 00-35 (Part 4) 'Natural Environments' (1986) suggests that wind speeds are unlikely to exceed 50 m/s anywhere on the seas, and combining these two factors leads to a *worst case* wind pressure of 0.5 kPa. It will be noted that this is at least an order of magnitude less than design pressures necessary to withstand green sea loading.

5 Wind speed varies considerably with height above sea level and the above speed is quoted at the standard height of 10 m. Curves or formulae for variation of wind speed with height are given in both the references in paragraph 4, but for practical purposes, unless a structure is particularly unusual, it is recommended that all above water structure is designed to a minimum pressure of 0.5 kPa *where no other more severe conditions apply.*

Inertial Loads

6 Inertial loads are those caused by the motion of heavy masses following the motion of the hull. Provided that the local accelerations are known, the estimation of inertial forces is straightforward (Force = Mass x Acceleration). However, estimation of accelerations is not directly possible until sufficient information is known about the form and mass properties of the hull to undertake a full seakeeping analysis. When such information is available it is usual to take the mean of the 1/10th highest accelerations which can be deduced from the relevant (heave, pitch or roll) response spectrum, that is

$$a_{1/10} = 2.55\sqrt{m_{4r}} \qquad (4.1)$$

where m_{4r} is the 4th moment of the response spectrum (to yield acceleration), to which must be added the acceleration due to gravity.

7 In the early stages of design when information about the ship is limited it is reasonable to assume simple harmonic motion, so that

$$F = 4\pi^2 \mathbf{m}\mathbf{d}\Theta/\mathbf{T}^2 \qquad (4.2)$$

where **m** = mass of item

d = distance from centre of rotation

Θ = amplitude of angular motion (zero to maximum, not double amplitude)

T = period of motion.

For example, if an item weighing 1.0 tonne is at 10 m from the centre of roll, for a roll period of 10 seconds and amplitude of 30° the force F is approximately 2 kN. Remember that the direct gravitational effect **mg** sin Θ must still be included, and that both must be resolved into the required direction before being added. As very much a first approximation, the total vertical force can be taken as 1.3**mg** on ships above about 5000 tonnes and 1.6**mg** on smaller ships.

8 Translational motions may be treated in the same way, but heave is likely to be the only significant one. For roll and pitch, centres of rotation may be taken as being in the waterline plane on the centreline, and at 2/3 of the ship's length from the fore end respectively. Although in theory the phase relationship between the various motions should be allowed for in combining them, in practice it is sufficient to assume that heave and pitch forces act together and are additive, while roll forces are independent. Heave and pitch forces will also act at the same time as the worst wave bending moments and, where relevant, the stresses due to all these loads should be added.

Thermal Loads

9 Moderate stresses in the structure can be caused by different temperatures in one part with respect to another part, or where materials with different coefficients of thermal expansion are used together. In warships the largest temperature variations will probably occur in the tropics where very high air temperatures in addition to solar radiation may lead to the upper deck being as much as 40°C hotter than the hull below water. In such circumstances stresses of the order of 75 MPa with curvatures of 10,000 m radius vertically and 50,000 m horizontally may be expected in steel hulls.

10 Because the highest stresses are due to *high* air temperatures it is unlikely that they will be combined with high wave bending moments. Nevertheless, significant thermal stresses may occur under storm conditions where the main strength deck is enclosed by a deckhouse. For example, in the Arctic exposed portions of the hull may be at sub-zero temperatures while enclosed parts of the strength deck, especially in way of accommodation or machinery spaces, may be as much as 30°C higher. If the superstructure is inset from the ship's side then the upper deck would be subject to stresses of the order of 40 MPa; tensile in the exposed outboard areas and compressive where enclosed. Keel stresses would be about 10 MPa in compression.

11 The effect of these longitudinal stresses in the deck and keel is not especially significant (provided that notch tough steel has been used as discussed in Chapter 5) as compressive stresses are always balanced by tensile stresses elsewhere. While areas in compression will shed load earlier suggesting a reduction in hull girder strength, the equivalent tensile stressed areas, if they also are subjected to primary compressive loading, will approach failure later and so there will be a degree of self compensation. In fact, there will be a small loss of strength as compensation is not total, and using typical collapse behaviour taken from Smith and Kirkwood (1977) it can be shown that a difference of 30°C across the ship between exposed and enclosed parts of the strength deck typically leads to a reduction of overall hull girder strength of about 5%. For any particular design, if such differential temperatures are likely in association with extreme wave bending moments, then an allowance needs to be made within any safety margin that is applied to overall ultimate strength. This is particularly the case if aluminium is used in the structure as its coefficient of thermal expansion differs from that of steel while at the same time its Young's modulus is considerably lower.

Ice Loading

12 The information on ice loading in the following paragraphs is derived from Canadian Naval data and their contribution is acknowledged. Large portions of the world's northern and

southern oceans are sometimes covered with floating ice. In winter, for example, the ice edge can extend as far south as 45°N off the East Coats of Canada and off Northern Japan; sea ice will also be found in between Greenland, Iceland and Norway. Ice conditions are highly variable in both spatial and temporal distribution. The magnitude and nature of ice loads experienced by a hull in contact with ice will be greatly affected by these and by the method of operation of the ship. It should be noted that intelligent strategic navigation using best available ice information can result in ice impact loads that are an order of magnitude less than those that may be experienced if piloting unaided in ice covered waters.

13 Design ice pressures published in various design codes reflect the variability in ice and ship operations. Shell plating design pressures can be as low as 90 kPa (American Bureau of Shipping (ABS) Ice Class C) for the stern region, which would only be appropriate for the lightest ice conditions of open pack less than 0.3 m thick. On the other hand, shell plating design pressures of 12 MPa are suggested for ships operating in the high Arctic ice (ABS Ice Class A5). Equivalent values determined from Lloyd's Rules range from 380 kPa (Lloyd's 100A1 - Ice Class 1C at the stern (for up to 0.4 m ice) to 12 MPa (Lloyd's proposed Ice Class AC3 (1984)) at the bow (highest ice strengthening for multi-year ice). Ice loads are very localised and peaked, and average pressure drops off rapidly with the area over which the load acts.

14 Ice loads will dominate the local structural design of a warship hull in all but the very lightest of ice conditions. Therefore, for an 'ice capable' ship, that is a ship that has some ice breaking capability as opposed to resistance to the impact of reasonably loose ice (as defined in paragraph 15), the stipulation of any ice capability will affect the structural design and weight. It is therefore important that the extent of ice operations expected and the ice conditions to be used in design, based on the projected area of operations, are defined clearly. The technology for the design of ice capable merchant vessels and specialised icebreaking ships is currently advancing rapidly and should be applied to the design of an ice capable warship. Rules from a number of Classification Societies (ABS, DnV, Lloyd's, amongst others) are available as a guide to the designer, but a prerequisite is to define the area of operations, the average and maximum speeds and the most probable type of ice to be encountered.

15 This being said, most warships will not be ice capable nor will need to be, but some might need a measure of ice strengthening. The most likely area of operations of these warships will be in the marginal ice zone, that is the area near the first year ice edge. The ice will generally be in the form of loose first year floes (not surviving the summer) or pans of new ice. In such conditions the ice loads will act very near the waterline and should not be combined with wave or other loads. While there can exist significant swell at the ice edge, which could impart motion to floes, this occurrence is not sufficiently likely to be considered as the design condition. The loads will result from striking individual floes, which may then fail in bending or be pushed aside. In both these cases the loads are very local, are oriented horizontally and are highly dependent on the velocity of impact and the shape of the hull at the impact point.

16 For ice edge operations, that is in open (less than 50% cover) first year ice up to 1 m thickness as described in paragraph 15 above, the following approach to ice strengthening using a structural 'ice belt' is recommended. This is nominally equivalent to Lloyd's Register Ice Class 1B, ABS Class B or Finnish/Swedish Class 1B.

a. The vertical extent of ice strengthening should be from 1.0 m above the deep waterline to 1.3 m below the operational light waterline.

b. The horizontal extent of ice strengthening in the bow region should extend from the stem to 1.0 m abaft the beginning of the parallel mid-body or the maximum waterline beam. The stern region should extend from 1.0 m forward of the after end of the parallel mid-body (or aft of the bow region) to the extreme stern including the transom. The mid-body region is the remaining portion of the parallel mid-body, if any.

17 The design pressure (P_i) in MPa in the ice belt is given by:

$$P_i = P_o\,(\Delta/1000)^{0.2} \qquad (4.3)$$

where Δ is the deep displacement in tonnes and P_o is a nominal ice pressure to be taken as 0.90 MPa in the bow region, 0.32 MPa at the mid-body and 0.20 MPa in the stern region.

18 The shell plating in the ice belt should then have a minimum thickness (t_i) given by:

$$t_i = aK_f\,[\sqrt{\{1 + (\alpha/3)^2\}} - \alpha/\sqrt{3}]\,.\,\sqrt{(P_i/\sigma_y)} \qquad (4.4)$$

where a is the frame spacing

α is the aspect ratio ($\leqslant 1.0$) of the plate panel between stiffeners

and K_f is a constant equal to 0.6 for transversely framed structure and 0.7 for longitudinally framed structure.

To the plate thickness so derived, an appropriate corrosion allowance should be added as discussed in Chapter 5.

19 Equation 4.4 allows a small amount of yield at the edge of the plate and a small amount of permanent set, using a method described by Hughes (1983), and the load is the equivalent to approximately 1.25 times the load to initiate yield. The amount of permanent set will depend on the plate span, aspect ratio, thickness and the boundary conditions. The steel specified for the ice belt should have a minimum guaranteed low temperature notch toughness as defined in Chapter 5.

20 Framing should be designed against a uniform pressure load of P_i over an area with a vertical extent of 0.25 m and a horizontal extent of 2.0 m at all points within the ice belt defined in paragraph 16. No yielding nor instability should be permitted in the framing.

21 Note that although this chapter is strictly concerned with *loading* and that *design* matters are covered in later chapters, it is more appropriate to discuss here plating and framing design criteria to withstand ice, rather than in Chapters 7 or 10, as the design is empirically associated with the load.

22 No special load case should be applied to appendages such as shaft brackets, because to enable them to withstand ice impact would severely impair their hydrodynamic performance. With prudent navigation (that is less than 7 knots, with constant ice observations and avoidance of old ice), the ice floes will not be pushed down and around the hull and so should not impinge on appendages.

23 Finally, for warships for which ice edge operations are not regularly expected it is not reasonable to specify additional ice strengthening. It is more appropriate to evaluate the ability of the structure to withstand ice loads, elastically and plastically, for a wide range of possible conditions. The designer is referred to Varsta (1983) and Coburn et al (1981) for discussion of ice loads in thin ice. It is recommended that for normal warship design this information be used to determine the maximum ice conditions and speed of operation which the vessel can accept based on the existing structural strength. It is then possible to specify the worst conditions in which the vessel should operate. Parameters such as ice thickness, concentration, floe size and ice decay state will need to be included in the description of the conditions.

24 The above considerations all apply to operations in sea ice. A separate loading condition can occur when ice forms on the deck and upper works of a ship due to freezing spray. The thickness of ice can build up to many centimetres if it is not removed, but is likely to cause a hazard first due to loss of stability rather than loss of strength. Nevertheless, if a ship is expected to operate for extended periods in areas where icing is likely, an allowance should be added to both the displacement (for calculation of wave bending moments) and to the pressure on exposed horizontal surfaces. Under such conditions a local mean thickness for grillage design of 100 mm of ice is probably adequate, while 50 mm should suffice over large areas.

Propulsion Train

25 The forces resulting from the propulsion train are due to the thrust transmitted from the propulsor through the shaft to the hull and from any out-of-balance in components of rotating machinery. The first will be established with some accuracy early in the design and is usually passed into the hull through a thrust block seating in the shaft line. If vectored thrust devices are used, such as Voith-Schneider propulsors, the thrust will be taken directly into the hull immediately above the device and can be in any direction. In either case, however, with the force known, the design of the seating will be straightforward; for further information on seating design see Chapter 12.

26 The problem of out-of-balance is less easily accounted for. Ideally there will be no out-of-balance but in practice it is difficult to ensure complete dynamic balance especially for large components such as the propeller. However, the forces will usually be small in relation to dead loads and inertial loads (except possibly for forces on shaft brackets should a propeller be damaged, as discussed in Chapter 12), and need only be considered for their contribution to vibration excitation. It is therefore important to ensure that local resonant frequencies near to machinery seatings are not close to any low harmonic of the excitation. For the whole hull or the mast only the propeller and shaft are likely to provide enough excitation energy and it is therefore

only necessary to ensure that the first three or four hull or mast modes do not lie close to a shaft or blade rate that is likely to be used for long periods during normal operation. Consideration must be given to torsional and horizontal bending modes as well as vertical bending, particularly if the hull has large cut-outs for machinery or weapon systems. Methods of vibration prediction and analysis are given in Chapter 14, but also see Todd (1961), the BSRA Ship Design Manual - Vibration (1981) and Ward (1983), and for an example of an analysis see Bishop et al (1986).

27 A final load not to be forgotten is that on the hull above the propeller due to uneven flow through the propeller disc, which is again a cyclic load at blade rate. Even if the propeller were operating in a uniform flow field a pressure variation would exist at the hull plating, but as long as the wake is reasonably uniform the forces will be of the same order as the other vibratory forces in the shaft line. However, if the propeller is operating in a markedly non-uniform wake, especially at high powers, there will be large pressure fluctuations on the hull nearby, as well as excessive cavitation forming at the back of the blades. As the cavitation bubbles collapse, further pressure fluctuations will be created and will cause large pressure pulses on anything in the vicinity, of a magnitude great enough not only to cause erosion but also to generate fatigue damage in the hull plating due to local forced vibration. It is not reasonable to design the hull against this effect as cavitation of this order is highly undesirable and much more significant effects will occur elsewhere more quickly. It is better to design the hull form to minimise the unevenness of the wake through the propeller, and an essential first requirement is a separation of at least 20% of propeller diameter between the hull and propeller disc.

Wheel Loads

28 Wheel loads result from vehicles on landing ships and RO-RO vessels as well as from aircraft, both fixed wing and helicopters, on most modern warships. Loads from vehicle wheels are most readily estimated as they consist only of dead weight and inertial loads. There is no specifically military requirement in this case and the rules of classification societies may therefore be followed, for example Part 3, Chapter 9 of Lloyd's Rules and Regulations. To use the method it is necessary to know the maximum axle load and the maximum individual tyre load taking account of vehicle cargo, if any. Either the scantling rules of the classification society may be used or, if preferred, direct calculation is permitted using an allowance for ship motion as described in paragraph 6-8 above. Provided that the worst case of vehicle wheel load and ship motion is taken then it is not necessary to combine vehicle wheel loads with any other load, unless there are large numbers of vehicles parked close together on the main strength deck. In such a case the load should be treated as a *continuous* pressure applied to the deck for deck design as described in Chapter 7.

29 Aircraft loads apply to both flight decks and hangar decks. The most severe load in a hangar will be the parking load which is related to the all-up-weight (AUW) of the aircraft; on the flight deck landing loads, related to the aircraft undercarriage oleo loads, will predominate except in areas which are dedicated to parking only. The principle to be applied for landing areas is that the aircraft structure will fail before the ship, and in particular that there is no risk of rupturing the deck and allowing burning fuel to run below should an aircraft crash. The maximum load that the undercarriage can apply is known as the oleo ultimate load (*not* the collapse load which refers to complete failure of the undercarriage). There is only a low probability, confirmed by landing trials and accident records, of the *ultimate* load being reached on a single wheel, even during a crash, and so the *collapse* load is of little relevance.

30 For aircraft landing therefore, the load that should be used is the greatest oleo ultimate load on any undercarriage that is likely to land on a particular part of the deck allowing for all likely aircraft and every wheel of each aircraft. To allow for dynamic effects this load should be factored up by 1.5 for stiffener and by 1.2 for plating, the different factors allowing for the difference in stiffness of the parts of the structure. The background to the derivation of these factors may be found in Jackson and Frieze (1980).

31 For parked aircraft the design load is based on the AUW distributed correctly between all the wheels. The load on each wheel should then be factored up by 1.75 for aircraft on an exposed flight deck, and by 1.6 for aircraft in a hangar. These factors include partial factors for motion (~1.25), tie-down restraint (~1.25) and windage (~1.1) where appropriate.

32 Once again these loads may be taken as being independent of all others and applied separately. The only possible exception might be if large numbers of aircraft are to be parked close to each other on a main strength deck, in which case the parking load should be treated as a continuous lateral load applied in conjunction with the primary in-plane load as for parked vehicles as discussed above.

Internal Deck Loads

33 Forces applied to decks by concentrated loads are described above (paragraph 6 et seq) and by the sea in Chapter 3. However, on internal decks there will be many loads due to equipment, personnel and other items which are too many to allow for individually and indeed may move about or be spread over an area. It is necessary to account for such items during design of internal decks and the simplest way is to use an equivalent pressure on the deck.

34 The first consideration in estimating the pressure is whether the structure will be subject to air or water testing as described in. Generally only tanks will be water tested and the pressure for the test will be derived from the height of the vent above the tank top with an allowance for surge effects and for the speed of filling; note that this pressure is applied *upwards* to the tank top. Otherwise the air test pressure required for leak testing is 13 kPa, which is equivalent to a depth of about 1.2 m of water on the deck and is fairly severe. This pressure is only likely to be exceeded in magazines and storerooms, where it may be possible to make a reasonable estimate of the load knowing what is to be kept in the compartment and spreading the mass over an appropriate area, with an allowance for additional stores in an emergency. If this information is not available then a pressure of 15-20 kPa should be allowed depending on the density of the stores.

35 In compartments which will not be subject to test pressure, the actual loads spread over the deck should be used if known, but for lack of better information it will be sufficient to allow conservatively for 10 kPa in workshops and offices, and 5 kPa in living spaces and passageways.

Docking and Slipping Loads

36 Docking a ship on blocks imposes very high vertical loads on the keel. Slipping a ship, on the other hand, involves drawing the ship out of the water on a cradle which applies inherently less load than docking because the ship is fully supported along its length and transversely at all times. However, as all ships may be expected to be docked at some time, it is necessary to design for the docking condition as the most severe one. Unfortunately, the response of the dock blocks to the load is highly variable and depends not only on the local weight of the ship, but also on the stiffness of the bottom structure, the stiffness of the blocks, the accuracy of alignment of the top of the blocks and the straightness of the keel. If all these details were known then an accurate analysis would be possible, but as most of them will clearly vary from docking to docking it is not a feasible design approach.

37 It is recommended therefore that a simplified method is used whereby a line load at the keel equal to W_c/L_c (W_c is the weight of the length, L_c of the ship between main transverse bulkheads) is distributed equally over the blocks in that section, so that for normal block spacing the load on the keel can be taken as a distributed line load. This will be adequate provided that blocks are set up and aligned correctly, so a small margin may be appropriate for poor block condition, spacing or positioning.

38 The only exception will be in the region of the after cut-up where there is a considerable overhang increasing the load at the after end of the keel. It is a reasonable assumption that the perturbation to the load due to the overhang will extend forward from the cut-up a distance equal to twice the length of the overhand (L_o) and will be distributed parabolically. It may then be shown that the mass (w) per unit length to be added at each dock block at x forward of the cut-up is given by:

$$w = (3W_o/4L_o^2)\,[(2L_o + 3L_G)\,x^2/L_o^2 - 2(3L_o + 4L_G)\,x/L_o + 4\,(L_o + L_G)] \tag{4.5}$$

for $0 \leqslant x \leqslant 2L_o$ and where W_o is the mass of the overhang with a centre of gravity distance L_G abaft the cut-up. The distribution is illustrated graphically in *Figure 4.1*.

39 The loads due to side blocks and docking shores are much more difficult to estimate, but for practical purposes it may be assumed that either the loads will be taken directly into transverse bulkheads in-plane or that the loads, when applied to the stiffened side structure, will be less than that allowed for during normal seaway loading. Consequently, there is no need for the designer to take account of these load sources unless there is a very unusual configuration of structure or of docking procedure, in which case experiments will be needed to establish an appropriate design load.

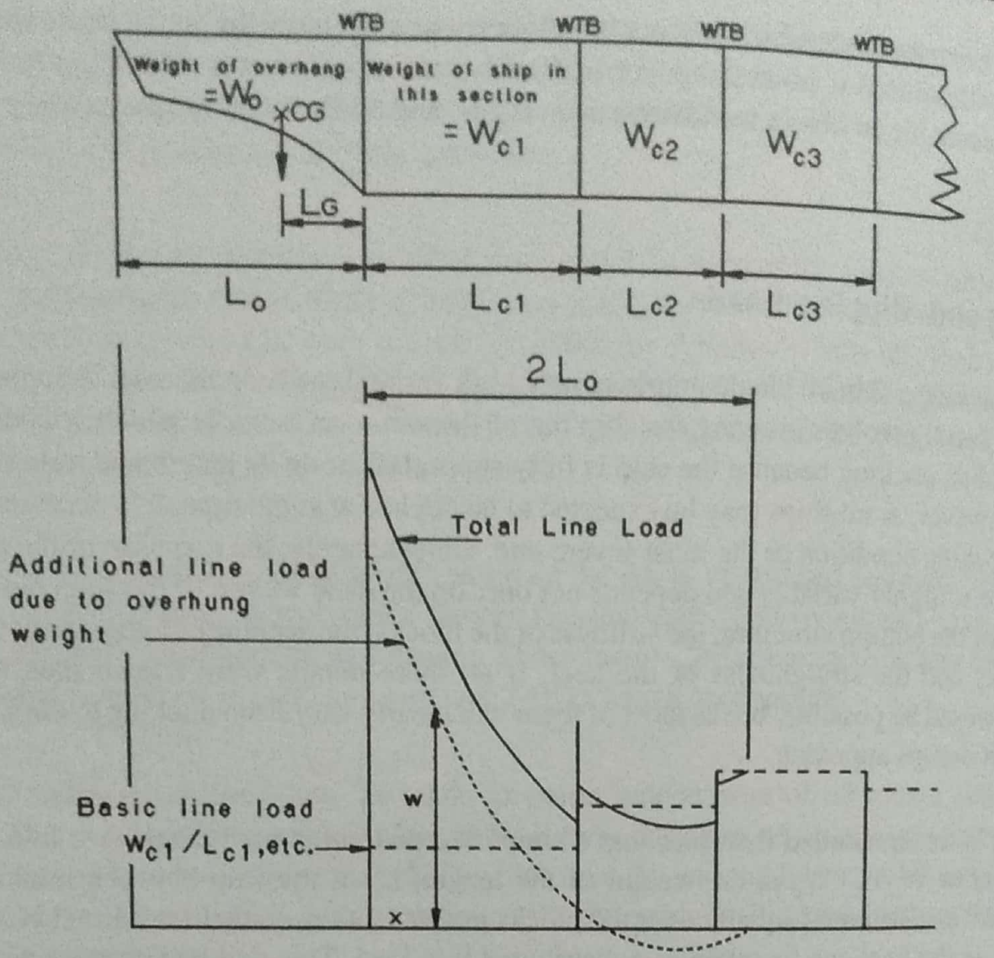

Figure 4.1 Calculation of Dock Block Loads in way of the After Cut-Up

Berthing Loads

40 Berthing loads are extremely variable depending, as they do, on the skill of the officer in command, the prevailing weather conditions and the resilience of the structure against which the ship is berthing. It is unrealistic to design ships' hulls to withstand unusually heavy berthing loads and a simple rule of thumb that has been developed over the years from ships which have suffered minor damage is that the structure within a 1.0 metre strip centred on the deep waterline should be able to withstand a line load of displacement/800 per metre length so as to avoid damage under normal conditions. However, if the ship is particularly large, or can have significantly different loading conditions, the 1.0 m strip will need to be widened depending on operating requirements.

Launching Loads

41 Bending stresses in the main hull girder due to launching are only moderate, typically of the order of 50 MPa for frigate sized ships. Loads should be checked by the shipbuilder using conventional analytical methods, for example as described by Rawson and Tupper (1976) but,

occurring only once in the life of a ship, they do not normally warrant special attention from the structural designer. However, the fore poppet should be carefully designed to ensure that the high loads to be transmitted in this area are applied to stiff structure in the fore end of the hull and, if adequate structure is not available, temporary stiffening may be necessary.

Grounding Loads

42 For the majority of ships grounding is an accident condition and does not directly affect the design of the structure. For vessels which are expected to ground as part of their normal operation, for example landing ships and some small boats, then the local loads and resulting bending moments applied to the hull at all relevant states of tide and bottom conditions need to be calculated. Methods may be found in most general naval architectural textbooks, for example again Rawson and Tupper (1976), but the problem usually refers to loss of stability during grounding. It is, however, a fairly simple matter to convert the forces so calculated to bending stresses on the hull. Local loads are less easily allowed for as the area over which the force is applied depends on the flexibility of the ship's bottom and type of surface on which grounding takes place. It is probably best to assume that all the load is spread evenly over a single panel of plate between stiffeners which is reasonably conservative without being undesignable. If the structure is made from a material other than steel then it will also be necessary to make an allowance for abrasion on the bottom, either by providing a tough surface or by thickening the plating to allow for some sacrificial loss.

Collisions

43 A collision is clearly another accident condition with which the structural designer rarely has to concern himself. However, there may be occasions when it is desirable to design a structure to withstand collisions, for example for nuclear powered ships. It is extremely difficult to predict the loads that will result from a collision as there are so many variables and most methods are based on estimates of the energy or momentum lost by the vessel being decelerated. Some research work was published in the 1960s in support of a few designs of nuclear merchant ships that were then being developed and a summary of design methodologies is presented by Van Mater and Giannotti (1979). More recently, however, with the increasing incidence of minor collisions between offshore support vessels and fixed platforms there has been an upsurge in research interest and the current position is described in the report of ISSC Committee II.2 - Dynamic Load Effects - (1988) and by Ronalds and Dowling (1988).

Towing Loads

44 Other than tugs, for which reference should be made to Lloyd's Rules, ships do not tow others as a matter of course but only in an emergency. As a general rule a tow should always be arranged so that the working load of the towing cable is less than the working load of the towing bollard or clench. However, conventional requirements are that the stress in the material of the towing clench when tested to the *proof* load of the towing cable should not be more than 80% of the yield stress. It is therefore appropriate in design to apply a load to the structure equivalent to 30% above the proof load of the cable or the cable breaking load, whichever is the least, thus ensuring that components fail notionally in a logical order. Remember also that the tow will be passed through a fairlead which will change the direction of pull, so the structure supporting the towing fairlead must be designed to the appropriate resultant load. The same philosophy may be applied to anchoring and mooring arrangements for the specified test loads.

45 Loads due to towed sonars are not generally significant as far as the primary structure is concerned, although it may be necessary to take account of unusual operating patterns when deciding the margins to be included in the estimation of design wave bending moments, especially for fatigue (see Chapters 3 and 13). The sonar cable winch will usually be designed to render at a much lower load than the breaking load of the cable, probably about half, so it will be sufficient to design the winch supporting structure and any associated fairleads to the breaking load of the cable.

Weapon Launching Loads

46 Loads resulting from weapon launch may include recoil effects, blast and missile efflux pressure and, in general, will be impulsive. These three types of load are estimated in different ways and will be covered in turn.

47 Gun and mortar recoil will be found generally in the documentation for the mounting. A typical force/time curve for a single shot is shown in *Figure 4.2* and the important point to note is that the force rises very rapidly to its maximum and decays almost as quickly. There is therefore a natural period of excitation which, for practical purposes, can be taken as four times the time to

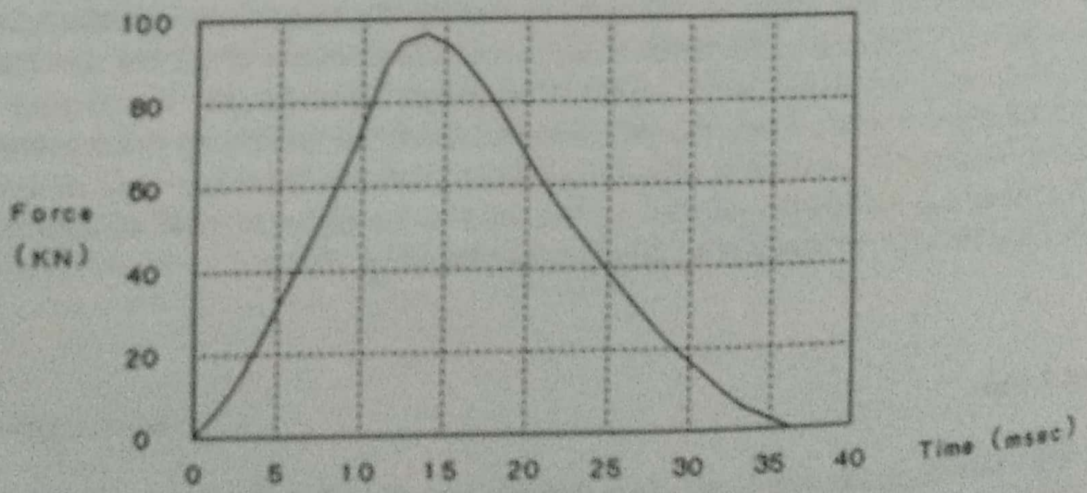

Figure 4.2 Typical Recoil Force/Time Curve

maximum force, which is superimposed on the excitation associated with the frequency of firing for multiple shots. In assessing the response therefore the load must be taken to be periodic and ideally a dynamic analysis of the supporting structure should be undertaken as will be discussed in Chapter 14. However, for initial design purposes it is sufficient to use an approximate 'dynamic load factor' which allows for any possible magnification of the exciting load due to resonance effects.

48 Provided that the natural frequency of the supporting structure is more than four times the firing rate and at least 50% higher than the frequency derived from the time to maximum force, then a dynamic load factor of 1.6 will be adequate for first estimates. If the gun is mounted immediately above a bulkhead then the structural resonant frequencies will be much higher and a dynamic load factor of 1.2 may be assumed. It is clear therefore that any support to a gun or mortar mounting must be substantial and provide a fairly high stiffness if unacceptable resonant responses are to be avoided. For further information on structural resonant frequencies see Chapter 14.

49 A final point to be remembered is that the line of action of the recoil force will depend on the azimuth and elevation of the gun. It is important therefore to choose the worst cases, which will usually be ahead and abeam and at 0° and maximum (or 90°) elevation, although checks should also be made at the 45° positions vertically and horizontally against the resolved in-plane and normal elements of the loads which occur simultaneously.

50 The load on the structure due to gun blast is in the form of a short-lived transient overpressure; values of this overpressure should be available in the documentation for the weapon as curves of pressure against distance from the gun muzzle. The pressure will act only for a time of the order of 10 ms so the structure, with a much higher natural response period, is unable to react to the full overpressure and it is sufficient to design to an equivalent static pressure of 25% of the peak blast pressure. Should blast pressure curves not be available then an approximation to the equivalent static design pressure can be found from

$$p_g = 2MPa\{(1 + \cos\vartheta)^2/(d/b)^{1.5}\} \tag{4.6}$$

where b is the bore of the gun and d is the distance from the muzzle of a point at which the pressure p_g is required on the structure on a line at angle ϑ to the centreline of the barrel.

51 Missile efflux blast loading can be predicted by considering the rate of change of momentum of the efflux where it strikes the structure under consideration. However, when calculating the equivalent design load allowance must also be made for the dynamic response of the structure, which leads to an undertaking of considerable complexity. For practical purposes therefore it is sufficient to design for the thrust averaged over a cone of semi-angle 3° and the resultant equivalent static pressure p_m may be found from

$$p_m = 1.5\ (T_m/A_i)\ \{\sin\alpha/(\sin\alpha + \tan\beta\cos\alpha)\} \tag{4.7}$$

where 1.5 is a dynamic load factor relating to variations in the efflux pressure rather than the structural response, T_m is the thrust acting at an angle α ($25° < \alpha < 90°$) to the structure over an area

A_i and ß is the efflux cone semi-angle. Both equations 4.6 and 4.7 are obtained from information not in open literature and are approximate but adequate for the purpose.

Loads Due to Underwater Explosions

52 The derivation of standards for design to resist underwater shock, both for structure and equipment, is based on shock trials carried out on old vessels at the end of their lives. The *structure* of none of these vessels had been designed specifically with underwater shock in mind (except that some later ships will have had locally improved bulkhead design as described in Chapter 9), and so the standards derived for *equipment* relate to structure with no explicit shock hardening. In general, except for some local details, the structure of the trials vessels withstood shock to a higher level than the equipment on board, and current equipment standards are still set at a somewhat lower level than the traditional hull resistance. Consequently there is no need to increase the shock resistance of the structure above that which had previously been achieved, or indeed to allow explicitly for shock loading during overall structural design. It is, however, possible for shock loading to cause premature tearing of structure with subsequent risk of flooding in local areas of high stress; the design of such details is discussed in Chapter 9 (for bulkheads) and Chapter 15.

53 An entirely separate phenomenon associated with underwater explosions is hull whipping, that is large amplitude resonant vibration of the hull in its lower modes excited by the shock pulse from the explosion. In steel ships it is difficult and probably not worthwhile to design to improve resistance to whipping, which is clearly related to the stiffness of the hull; lower amplitudes and strains (and higher frequencies) can only be achieved by a stiffer and heavier structure than is required for wave loading and may only be justified in a ship which is expected, in normal operation, to be subjected to explosive loading, such as a mine countermeasures vessel. If, however, the hull is built of a low stiffness material such as glass reinforced plastic then it will be desirable to estimate the explosion severity necessary to damage the hull, compare it with the operational requirement and stiffen the hull if appropriate (see paragraph 36 of Chapter 15). Some guidance on the sensitivity of hull resonances to changes in stiffness can be found in Chalmers (1988(a)), but advice is best sought from explosives specialists.

Loading Due to Air-Blast

54 Air-blast pressure loading on warship structures can result from several sources, including nuclear weapons, conventional high explosives (HE) and fuel air explosives (FAE). The following cases are of relevance to design of blast resistant structures:

a. Far-field nuclear airburst explosions.

b. Medium standoff FAE explosions.

c. Close external HE explosions.

d. Internal HE explosions, either from hostile weapons or detonation of a ship's own ammunition.

55 These sources create different loading phenomena, but generally (a) and (b) above produce similar effects, varying primarily only in direction of blast loading, and can be combined as distant external air-blast loading. Items (c) and (d) can be treated as internal or close external blast, where fragmentation effects should also be considered.

Distant External Air-Blast

56 As stated above, both nuclear and FAE explosions produce similar phenomena. In practice the nuclear explosion will be the most demanding case, and design to withstand nuclear air-blast will result also in some hardening against FAE weapons. A far-field nuclear weapon detonation produces two main effects, which are described in detail in Biggs (1964) and Glasstone and Dolan (1977). These phases are:

a. The diffraction loading phase, during which the incident blast front reflects and diffracts around the target causing intense impulsive loading

and

b. The drag phase, during which the air behind the blast front causes high, long duration aerodynamic forces on the target.

Knowledge of the phenomena can be used to construct the load-time history on the above water structure of the vessel for subsequent simplified dynamic response analysis in the early stages of design.

57 The following is a distillation for naval structures of the classical approach to air-blast as detailed in the above references. As such, this reflects the current two-dimensional state of blast load analysis. Methods of computer aided load prediction against three-dimensional ship topside configurations are under development at the Defence Research Agency, Dunfermline and the Defence Research Establishment, Suffield, Canada, and these techniques should be available in the near future.

58 When an explosion occurs in air, a spherical shock front propagates from the point of burst and carries with it a peak overpressure (the free field overpressure P_{so}) which decays behind the front to a pressure p(t) given by equation 4.8

$$p(t) = P_{so}\,(1 - t/t_d^+)\,\exp(-at/t_d^+) \tag{4.8}$$

where t_d^+ is the positive duration of the pulse (of the order of 2 seconds) as shown in Figure 4.3 and a is the decay coefficient which varies between 0.1 and 10 but is about 1 for a nuclear blast an 3 for a fuel-air blast. The positive duration of the pulse is a complex parameter and further information would need to be obtained from specialists.

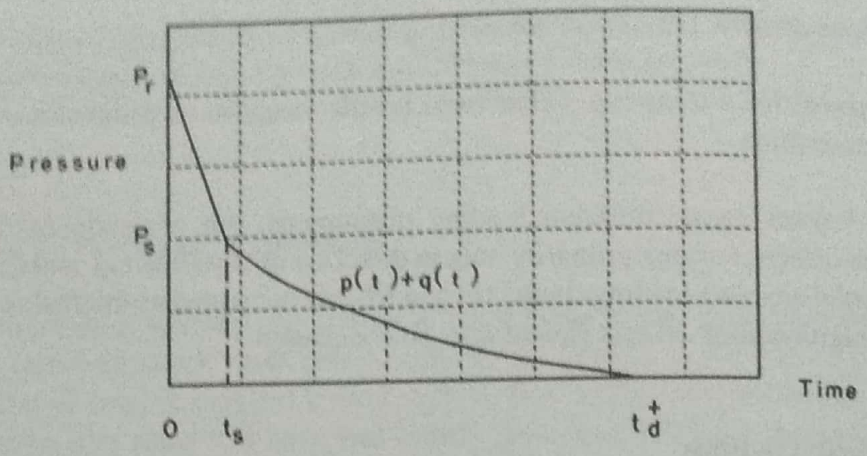

Figure 4.3 Average Front Face Air-Blast Loading on Closed Box-Like Structure

59 At the stand-off ranges of interest in warship design the shock front is seen as a planar wave front. As the blast front impinges on objects in its path, reflection of the wave causes a short term magnification of the peak pressure illustrated in *Figure 4.3* taken from Glasstone and Dolan (1977). The duration t_s of the peak is given in equation 4.9

(4.9) $t_s = 3S_c/U$

where U is the velocity of the shock front given by

(4.9a) $U = U_o \sqrt{[1 + 6P_{so}/7P_o]}$

and S_c is the height or half breadth of the deckhouse, whichever is the smaller, U_o is the ambient speed of sound, 340 m/s, and P_o is the ambient atmospheric pressure. After time t_s the pressure drops to the stagnation pressure P_s given by

(4.10) $P_s = p(t_s) + C_D q(t_s)$

where p(t) and q(t) are defined by equations 4.8 and 4.12 respectively, and C_D is the drag coefficient of the structure, discussed for different structural arrangements in Chapter 15.

60 The resulting peak reflected pressure P_r against solid objects is between two and eight times the value of P_{so} for a shock front striking a surface normal to its direction of travel. The actual magnification is dependent upon the value of the incident overpressure and the stiffness of the structure. Clearly a less stiff structure (that is, a structure with a natural period of the order four times the duration of the peak) will significantly reduce the intensity of the reflected pressure by absorbing the shock wave energy as deflection. However, the ship structures of interest are usually of higher natural frequency than this and therefore the full intensity of the reflected energy, as produced against a solid object, should be imposed as a structural load. This reflected pressure peak dominates the early stages of air-blast loading and is given by equation 4.11

(4.11) $P_r = 2P_{so}\,(7P_o + 4P_{so})/(7P_o + P_{so})$

61 For non-normal incidence of the blast front, the reflected pressure is reduced, but as this reduction is not significant for angles of inclination from the normal to the target surface of less than 35° this effect should be ignored for conventional superstructure configurations.

62 Concurrently the high speed flow of air behind the shock front causes a dynamic pressure q(t) given by equation 4.12 which creates a drag pressure on objects equal to $C_D q(t)$.

(4.12) $$q(t) = Q\,(1 - t/t_d^+)^2 \exp(-2at/t_d^+)$$

where Q is the peak dynamic pressure given by

(4.12a) $$Q = 2.5P_{so}^2/(7P_o + P_{so})$$

This dynamic pressure also decays behind the front, but is of much longer duration than the reflected pressure peak, resulting in an overall pressure time history of the shape shown in *Figure 4.3*, for the front face of a superstructure block. It should be noted that drag on the structure is present as soon as the blast front strikes, but only becomes dominant when the initial reflected peak has passed.

63 The side and top faces of a structure do not see the reflected pressure peak and therefore their pressure time histories for a structure of length L_s (the length of the deckhouse in the direction of the shock front propagation) are as shown in *Figure 4.4* also taken from Glasstone and Dolan (1977). It should be noted that for the top and sides of a structure the drag coefficient is negative, reducing the total load in the drag phase. However, as any face of a superstructure block may be facing the blast, all deckhouse sides should be examined against the more stringent case of front loading, whereas deckhouse tops should be loaded as in *Figure 4.4*.

64 For the rear face the loading is similar, except that there is a time delay of $(L_s + 4S_c)/U$ as the pressure builds up. This gives the pressure time history shown in *Figure 4.5* from the same reference. Again the drag coefficient is negative on the rear face.

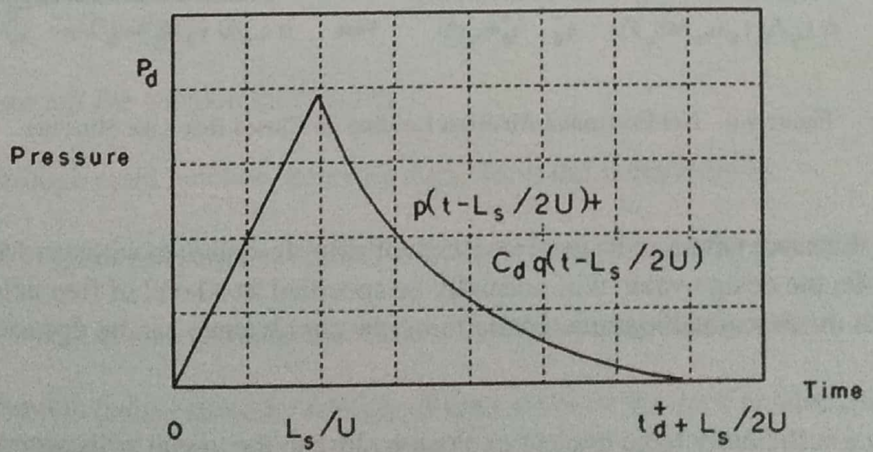

Figure 4.4 Average Side and Top Air-Blast Loading on Closed Box-Like Structure

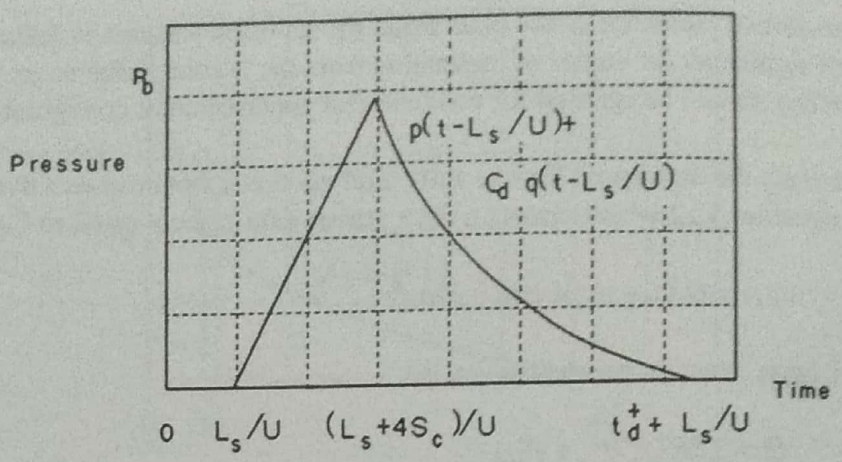

Figure 4.5 Average Back Face Air-Blast Loading on Closed Box-Like Structure

65 Thus, where a shock front crosses a superstructure block the net force time history on the portal frame of the deckhouse needs to be constructed from the sum of the forces on the front and rear faces, taking into account the time taken for the blast front to cross the structure as shown in *Figure 4.6*. In this case the rear face of the deckhouse, which does not see the high reflected pressures, tends to reduce the frame loading in the drag phase.

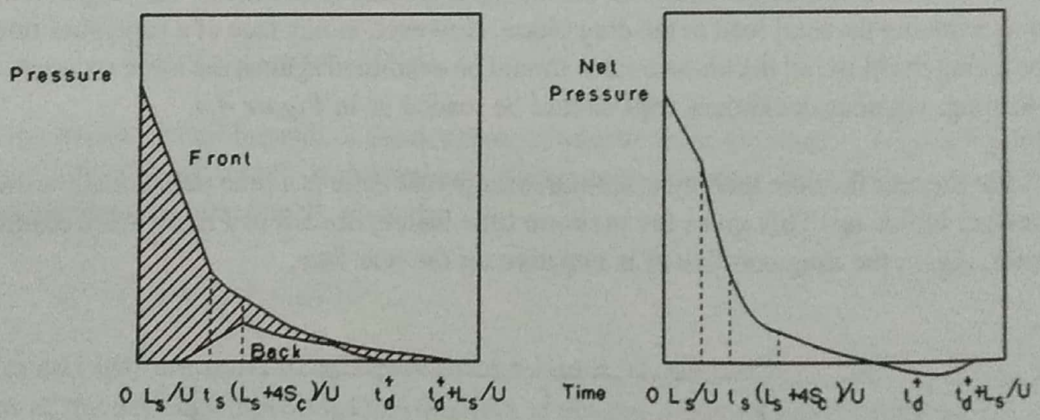

Figure 4.6 Net Horizontal Air-Blast Loading on Closed Box-Like Structure

66 Whilst the exact values to be used in design of ship structures to withstand air-blast are not openly available, the design value will normally be specified as a level of free field overpressure P_{so} from which the associated parameters defining the shock wave can be deduced as discussed above.

67 Clearly a sufficiently large nuclear explosion close to the vessel will cause capsize as well as the risk of excessive radiation dosage resulting in significant crew degradation within hours. In addition, survivability under nuclear air-blast is not usually a primary design aim of the vessel.

Therefore the design air-blast levels for surface warships tend to be limited to what is practically achievable without resulting in an unbalanced design. Nevertheless, for a superstructure block designed to be ineffective with regard to longitudinal bending (see Chapter 8), the air-blast load will be the dominant structural design parameter.

68 Similarly, capability against air-blast can be provided in two tiers. Firstly elastic design can be imposed against limited overpressure to ensure combat capability with undisturbed alignment of all major weapons and sensors. The second level uses plastic design in which loss of combat capability is accepted, and the aim is to maintain citadel integrity while allowing some structural deformation.

69 For nuclear air-blast the associated thermal energy effects also need to be considered. To do this the designer needs to make assumptions regarding weapon yield and standoff to achieve a realistic combination of free field overpressure and thermal load (normally expressed in J/cm). In practice the thermal load is only likely to be of significance where aluminium or fibre reinforced plastic structure is subject to nuclear blast.

Internal or Close External Air-Blast

70 Whilst air-blast damage may also be significant for close detonated high explosive weapons, the combined effects of fragmentation and air-blast will tend to complicate analysis of the effect on ship structural integrity. For simplicity these problems are typically separated such that damage from fragmentation is assessed in terms of system loss using vulnerability assessment tools (for example, REVS User Guide (1987)), and blast damage is assessed using methods such as those described by Baker (1973, 1980).

71 Thus for internal blast the analysis is sometimes simplified to the extent that a zone of devastation is defined, where structure in way of the detonation is removed and large deformation of surrounding structure occurs. Outside the area affected by fragments, which is usually completely destroyed, the remaining damage is due to a rapid build-up of internal pressure. This is usually vented overboard or into adjacent compartments via the weakest path of resistance which is usually in the order:

a. Through the weapon entry hole;

b. Through open hatches, doors or ducts from the compartment;

c. Through larger fragment holes, which are then enlarged by the blast;

d. Through closed watertight doors and hatches being blown open;

e. Through holes caused by rupture of fillet welds at the deck connections of watertight bulkheads;

f. Through failure of full penetration welds in the shell.

Decks, being continuous, tend to absorb much of the overpressure by large plastic deformation, provided that they have not suffered significant fragment damage. The designer must then analyse the residual strength of the remaining structure in a moderate sea state in which it is intended that the ship will be able to return to port. The procedure for this will be the same as for the intact ship, but with the wave loading reduced to match the lower sea state.

72 Clearly the extent of damage may be reduced through the use of blow-off plates, blast resistant watertight doors and hatches and the incorporation of more ductile structural connections in the design. However, in general these types of solution are still in the research domain and there is little, if any, information currently available to help the designer.

Lifting Loads

73 Some small vessels may be designed to be lifted in slings or from lifting eyes. This is a severe load condition, but is easily calculated using simple beam theory from the position of the lifting points and the vessel condition (usually a light displacement). Unlike in the buoyant state, however, the shear force at the lifting points may dominate the structural design rather than the maximum bending moment. Another point to be noted is that, if lifting eyes are used, then these and the surrounding structure to which they are attached will probably need to be made of high strength notch tough steel because of the local stress concentrations and possible low temperatures during lifting.